Optimized Main Bearings Oil Groove for Reduced Oil Pump Losses

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ABSTRACT

The main focus of ROTA 2030 program is to identify how to enable further fuel savings giving the efforts implemented by Inovar-Auto, e.g. at crankshaft area.

In a way to reduce friction losses and fuel consumption, OEMs are adopting features like variable oil pumps, new crankshaft oil drilling (e.g. one main bearing feeding two connecting rod bearings) and new configuration for the main bearings oil grooves. Usually a full groove (180°) is specified for the upper main bearing once it delivers the necessary oil flow and pressure to the connecting rod bearings with a cost effective manufacturing process. By adopting a partial groove configuration, which can typically vary from 90° to 170°, the overall oil flow to the bearings can be reduced once side flow is reduced. Such design solution enables the oil pump to work more efficiently specially when in a variable flow design. The challenge remains on keeping Minimum Oil Film Thickness (MOFT) safe to ensure robustness and durability. However, the adoption of partial grooves increases manufacturing costs considerably.

Using CFD simulation and dynamometer engine test, a new optimized main bearing oil groove was developed, which combines the reduced flow of the partial oil groove with the manufacturability of a full groove.

INTRODUCTION

OEM´s are putting more efforts towards engine efficiency increase in the past few years. The reason for that is known: besides reduced fuel consumption driven by customers, tax reduction given to more efficient engines can also improve competitiveness [1]. Such incentives are linked to programs like ROTA 2030. With that, areas beyond traditional engine calibration and PCU optimization are being explored further.

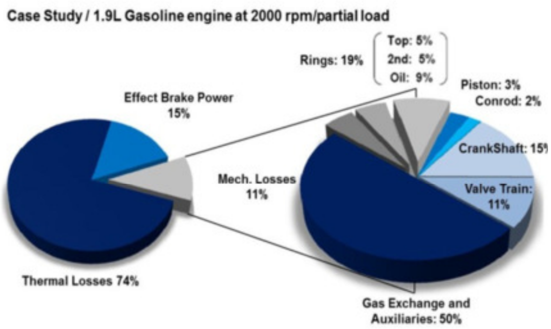

Figure 1: Energy distribution on light vehicle ICE [1].

That is the case of crankshafts and bearings within the ICE. As shown in figure 1, the crankshaft represents around 1.65% of mechanical losses and OEM´s are more frequently adopting configurations that reduce the oil flow to the crankshaft area. In some cases the oil flow is reduced by 50% or more [2] in a way to adopt smaller oil pumps reducing not only mechanical losses but also hydrodynamic losses once less oil is available to the moving parts. Figure 2 shows the typical main bearings features.

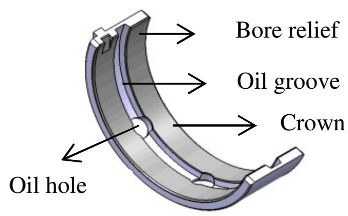

Figure 2: Main bearing design features

At usual crankshaft configuration, the main oil gallery supplies oil to the journals which finally supplies oil to the connecting rod pin. In this case, one main journal supply oil to one connecting rod pin, called "1 to 1" feed type, as shown in figure 3 and 4.

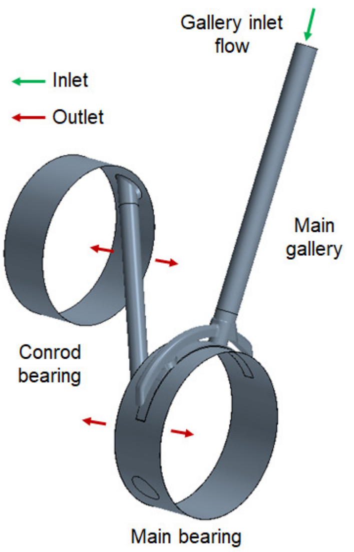

Figure 3: "1 to 1" feed type.

In configurations where reduced oil flow is desired, one main journal supplies oil to two connecting rod pins, called the "1 to 2" feed type, which is becoming more common. This configuration imposes a new working condition to the bearings that will work under a more challenging oil film thickness and potentially higher working temperatures [3].

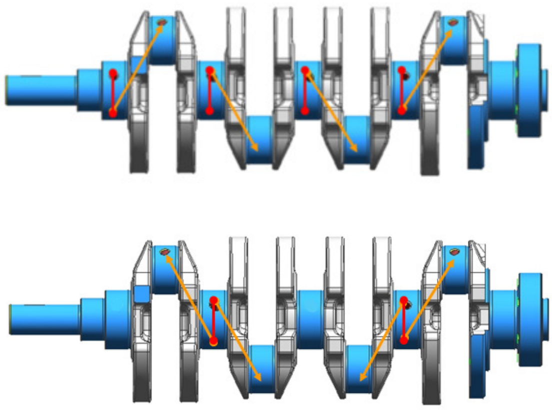

Figure 4: Example of "1 to 1" oil feed configuration (top image) and "1 to 2" oil feed configuration (bottom image)

MAIN BEARINGS DESIGN OPTIMIZATION

The main bearings have an important role in distributing the oil for the connecting rod bearings. Their groove configuration directly impacts on oil flow and pressure supply. To contribute with the oil flow reduction coming from the crankshaft optimization and more than that, to avoid issues with scuffing and wear increase, the bearings design had to change, especially the main bearings design.

The usual and simple configuration of the oil grooves on the main bearings might not be the best option for this new environment. With the reduced amount of oil available, the usual 180° oil groove configuration might not be sufficient to build a solid oil film for both the main and the connecting rod bearings.

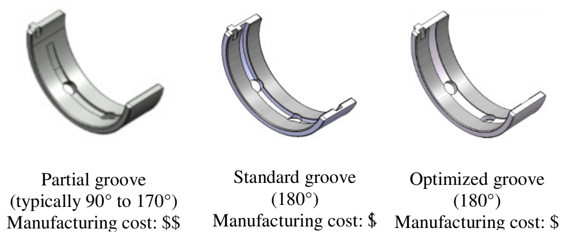

Figure 5: Typical groove designs (left hand side and center) and optimized design (right hand side)

The usual design alternative to reduce oil leakage and maintain adequate oil feed to the connecting rod bearings is to reduce the main bearings oil groove extent (typically varying from 90° to 170°) as showed in figure 5. This helps to direct the oil flow to the connecting rod bearings once the oil path from the main bearings crown toward the bore relief becomes more difficult.

Besides oil grooves, other bearings features like eccentricity, bore relief and assembly clearance can be explored to reduce oil flow. However the tradeoff between oil flow reduction and safe oil film working conditions might be difficult to achieve by changing these parameters, once it involves the overall system robustness to scuffing (increasing temperatures from low clearances for instance) and wear (reduced oil film thickness).

Traditional oil groove configurations like 180° extent used to work well and keep manufacturability and part costs under control for the traditional crankshaft oil drilling configurations. Both the conventional and the modern groove milling process like BFP – Bearings Finished at

Press – are possible to use with this design. On the BFP process the oil groove is milled on the bearing strip as demonstrated in figure 6, reducing operational costs.

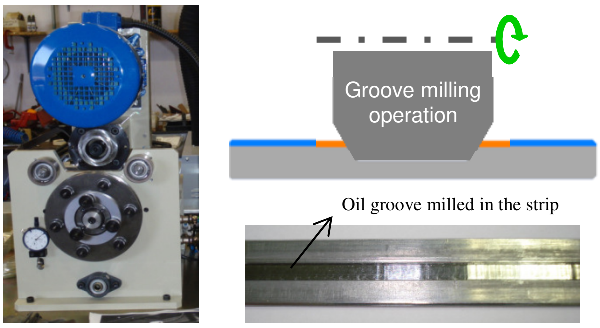

Figure 6: BFP groove milling operation done directly in the bearing strip, before forming.

However when a partial oil groove configuration is adopted to cope with the reduced oil flow, manufacturing costs increases once a more dedicated machining operation is necessary. It is possible to mill the partial oil groove using for example the BFP process, however this might also increase costs and reduce the line output as the oil groove milling station will become more complex. Increasing the BFP process complexity goes in the wrong direction once this process was developed to be lean to operate and a cost effective manufacturing process.

COST EFFECTIVE IMPROVED DESIGN

A solution for this issue was to develop an optimized oil groove geometry that holds both the advantages of partial groove design and the manufacturability of a BPF process. In order to understand the optimum main bearing oil groove design - keeping the 180° extent - that reduces oil flow to the level of partial groove configuration without jeopardizing other bearings properties like seizure and wear resistance, the CFD simulation was used.

NUMERICAL SIMULATION

Several designs were assessed varying the groove geometry. Six of them were selected to be simulated by CFD. The main groove dimensions and the difference among the designs are explained by figure 7 and table 1.

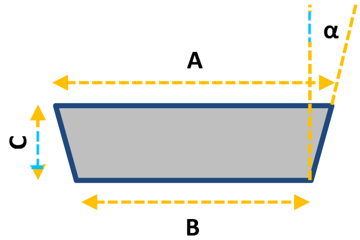

Figure 7: Oil groove cross section example and main dimensional parameters

Table 1: Oil groove designs simulated

	Oil groove dimensions			
CFD designs simulated	A (width at top)	B (width at bottom)	C (depth)	α (wall angle)
Design #1 Partial groove	Reference	Reference	Reference	Reference
Design #2 Standard 180°	Wider	= Reference	Deeper	= Reference
Design #3 Optimized 180°	Wider	Wider	Shallower	Sharper
Design #4 Optimized 180°	Narrower	Narrower +	Deeper	Sharper
Design #5 Optimized 180°	Wider	Wider +	Shallower +	Sharper
Design #6 Optimized 180°	Narrower	Narrower	Shallower +	Sharper

The CFD simulation indicated which oil groove design brought the desired oil flow reduction (results in figure 8) keeping adequate oil feed pressure to the connecting rod bearings (results in figure 9) in association with the lean manufacturing process.

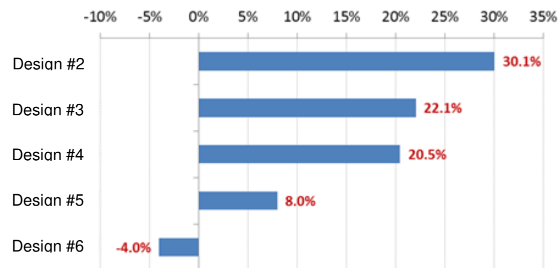

Figure 8: CFD main bearings oil flow results relative to partial groove (positive values = worse than reference).

As expected most of the proposed 180° groove configuration increased the oil side flow when compared to the reference (partial groove). The only exception applies to design #6 where 4% less oil flow was experienced. This design however brought an undesired effect of oil flow fluctuation inside the oil gallery, which can induce to air bubble formation in the oil, leading to a potential cavitation occurrence [4].

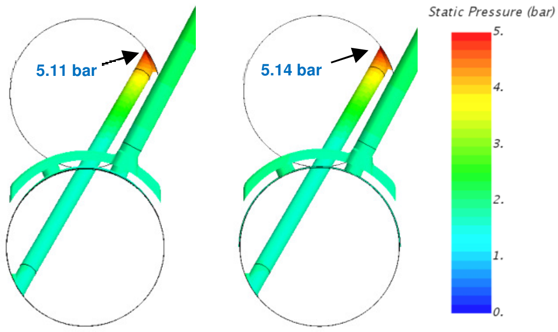

Figure 9: CFD oil static pressure to the connecting rods.

As a result, although design #5 increases the oil flow in 8% compared to the reference, it was chosen to be further explored and tested as being the one with the best balance between oil flow and other important parameters like oil feed pressure to the connecting rods (figure 9).

Prototypes were manufactured using the BFP process following the design #5 simulated by CFD, with the groove milled in the bearing strip, respecting the process capacity. In order to confirm the expected benefit given from this design, the main oil gallery of a four cylinder naturally aspirated engine with a variable oil pump was instrumented. After that a durability test was proposed on SI turbocharged flex fuel direct injection engine.

ENGINE TESTS

After being validated by CFD simulation the oil groove design #5 was tested on the dynamometer cell. The dynamometer cell is equipped with several controls to keep the temperature of fuel, intake air, coolant and lubricant oil under control.

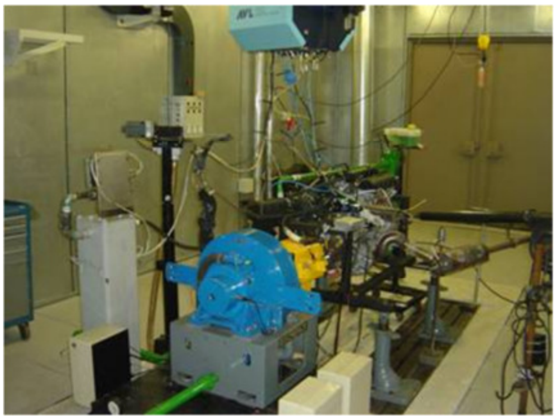

Figure 10: Dynamometer cell

A pressure transducer was installed in the engine main oil gallery with the purpose of evaluating the impact of the main bearings groove design in the system. At this same point a thermocouple was installed to control the oil temperature.

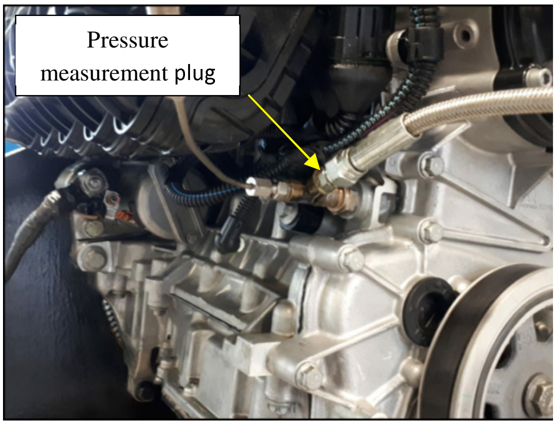

Figure 11: Oil gallery pressure acquisition location

OIL PRESSURE MEASUREMENT ENGINE TEST

The main oil gallery pressure was measured based on the engine performance curve protocol. Both the partial groove (reference) and the optimized groove were measured. The speed range was from 1500 rpm to 6500 rpm in full load condition with steps of 250 rpm. For each step the stabilization time was 180 seconds, thus ensuring that the engine parameters were stable for the measurement. For each main bearing design evaluated two performance curves were run and the average calculated.

Table 2: Test protocol details

Performance Curve WOT				
Step	Rotation (rpm)	Load (%)	Stabilization time (s)	Measurement time (s)
1,0	6250	100,0	180,0	60,0
2,0	6000	100,0	180,0	60,0
3,0	5750	100,0	180,0	60,0
4,0	5500	100,0	180,0	60,0
5,0	5250	100,0	180,0	60,0
6,0	5000	100,0	180,0	60,0
7,0	4750	100,0	180,0	60,0
8,0	4500	100,0	180,0	60,0
9,0	4250	100,0	180,0	60,0
10,0	4000	100,0	180,0	60,0
11,0	3750	100,0	180,0	60,0
12,0	3500	100,0	180,0	60,0
13,0	3250	100,0	180,0	60,0
14,0	3000	100,0	180,0	60,0
15,0	2750	100,0	180,0	60,0
16,0	2500	100,0	180,0	60,0
17,0	2250	100,0	180,0	60,0
18,0	2000	100,0	180,0	60,0
19,0	1750	100,0	180,0	60,0
20,0	1500	100,0	180,0	60,0

Table 3: Engine used for the main gallery oil pressure curve measurements.

Engine information	
Charging	Naturally aspirated
Number of cylinders	4
PCP	95 bar
Specific power output	60 kW/l
Specific torque output	105Nm/l

As anticipated by CFD simulation, it was expected that the oil gallery pressure would be slightly reduced using the optimized main bearing oil groove design (180°) compared to the partial groove. The figure 12 shows that the reduction was relatively lower (5% less) and at the speed range of 1500 rpm to 3000 rpm only. From 3250 rpm up to 4500 rpm both pressure curves overlapped. Finally, between 4500 rpm and 6250 rpm, a pressure increase was noted for the optimized groove configuration, including a delta greater than that saw at lower speeds. This effect, although not expected, is thought to be consequence of the variable flow that the oil pump imposes. Besides that not only the main bearings oil groove was optimized but also some structural and dimensional parameters. This helped minimizing the deformation imposed by the main bearing housing at high loads and speeds.

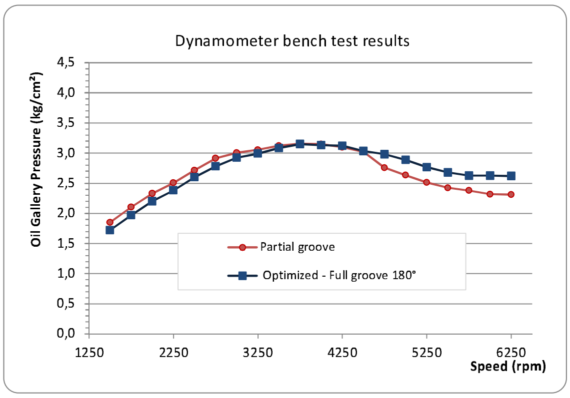

Figure 12: Oil gallery pressure results

DURABILITY ENGINE TEST

Near 600 hours were run on a SI turbocharged direct injection engine in order to assess if the visual aspect and wear performance were changed compared to historical results.

Table 4: Engine used for the durability test.

Engine information	
Charging	Turbocharged
Number of cylinders	4
PCP	125 bar
Specific power output	103 kW/l
Specific torque output	203Nm/l

No impact on wear was observed and all parts concluded the test successfully and without scuffing signs as the visual aspect shown in figure 13 demonstrate.

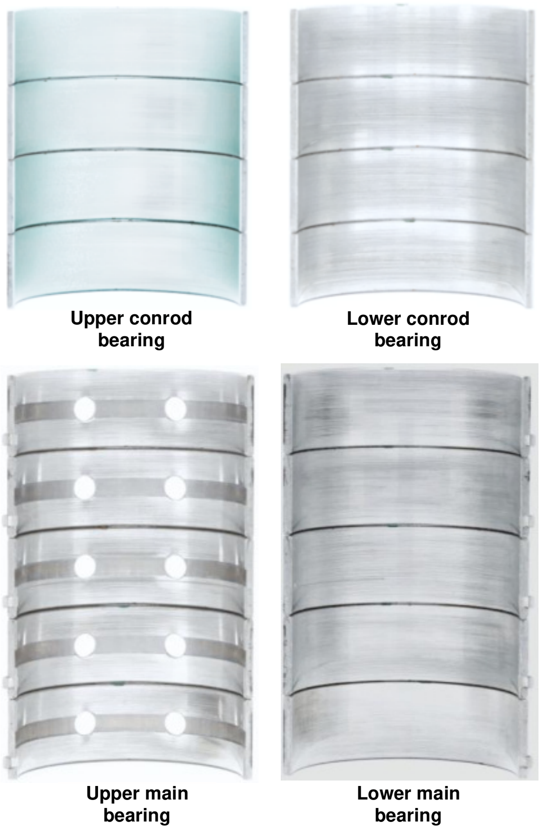

Figure 13: Visual aspect of tested bearings

CONCLUSION

The optimized oil groove design increased the oil gallery pressure by 2% on average in the measured cycle. At high speeds, where more oil is pumped to the lubrication system, even higher oil pressure was experienced by the system (up to +13%) compared to the partial groove design. This is beneficial once at high speed regimes the engine block – especially the Aluminum ones - trends to deform more due to combustion pressures and inertia forces, therefore having a higher oil pressure allows for a better heat removal from the system and solid oil supply to the connecting rod bearings.

The connecting rod bearing performance was not influenced by the new groove design and demonstrated adequate visual aspect and wear after test.

Durability engine tests have proven that this design did not reduce the overall bearings performance. The first SOP using this design will take place in 2021 on a high volume OEM.

One potential approach is to keep the same system pressure as the partial groove delivers, allowing the oil pump to be redesigned to reduce mechanical losses.

The use of an optimized 180° groove extent allowed the manufacturing costs to remain in the range of the traditional 180° groove configuration and provided a step change in costs compared to the partial groove, once the BFP process could be kept.

ABBREVIATIONS

OEM = Original Equipment Manufacturer

CFD = Computer Fluid Dynamics

PCU = Powel Cell Unit

ICE = Internal Combustion Engine

SOP = Start of Production

SI = Spark Ignition

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